

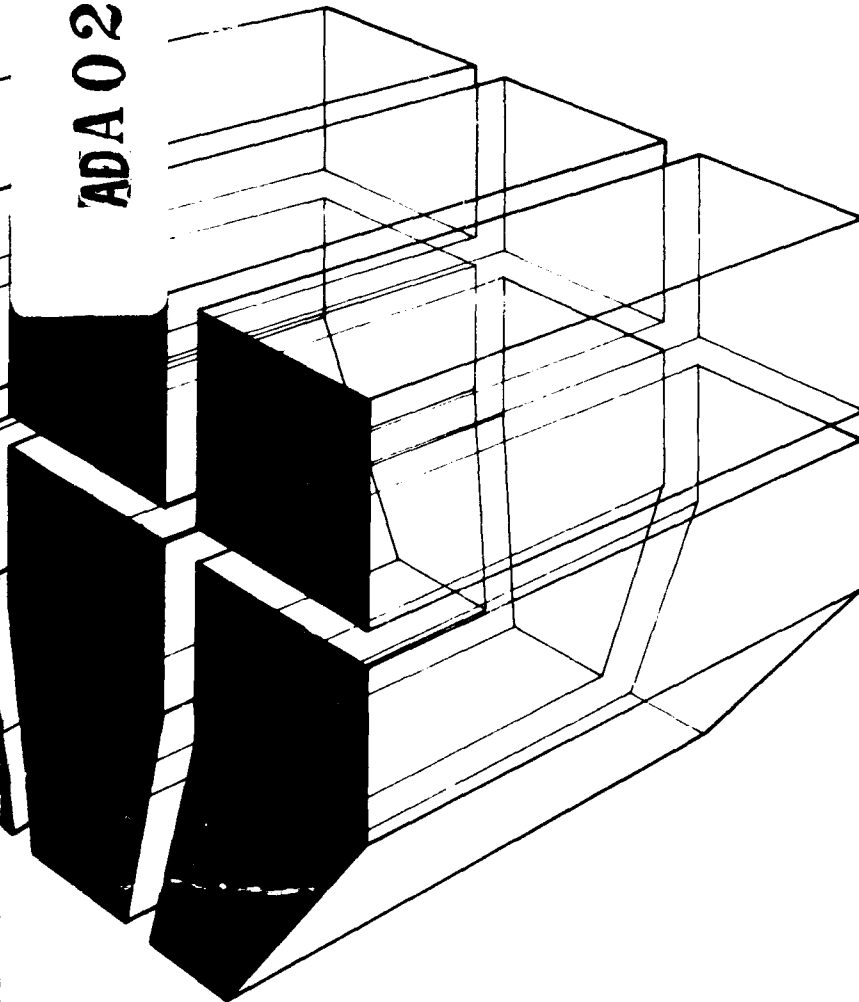
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August 1976

FAILURE ANALYSIS OF OZARK, ARKANSAS,
POWER PLANT SOCKET-HEAD CAP SCREWS

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by
E. P. Cox



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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) This research analyzed the socket-head cap screws which failed in the turbines of the Ozark Power Plant, Ozark, AR. The cause of failure was found to be fatigue in improperly heat-treated cap screws. The reduced fatigue strength of the cap screws was further aggravated by the presence of a sharp fillet where the shaft and head joined and by the nonuniform preloads imposed on the cap screws during installation.		

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Several procedures are recommended to improve the service life of the turbine cap screws:

1. Insure that all cap screws or studs are heat-treated in the following manner:
 - a) Provide full normalizing heat treatment (after forging) at 1600°F (870°C).
 - b) Austenitize for 2 to 3 hr at 1550°F (843°C), and quickly oil quench.
 - c. Temper at 800°F (423°C) to a hardness of 38 to 42 R_c (approximately 3½ hr).
2. Use rolled rather than machined threads for both cap screws and studs. The threads should be rolled on after heat treatment.
3. Keep cap screw and stud hardness below 44 R_c to prevent stress corrosion cracking.
4. Machine cap screw and stud/nut seats to insure good alignment and perpendicularity; this will eliminate bending stresses.
5. In the future, keep the threaded portion of cap screws not closer than 1 in. (2.54 cm) to the head. Similarly, the fillet should be increased to a minimum of 0.5 in. (1.27 cm) radius and locally shot-peened.
6. The preload used when installing the cap screws or stud/nut arrangement should be between 15 and 20 ksi (117 to 138 MPa) and should be imposed equally on all cap screws or stud/nut arrangements used in the turbine.

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FOREWORD

This investigation was conducted by the U.S. Army Construction Engineering Research Laboratory (CERL) for the Corps of Engineers Little Rock District, under Inter-Army Order 76-106. CERL personnel directly concerned with the study were Ms. J. Scott and Mr. E. Cox of the CERL Metallurgy Branch (MSM), Materials and Science Division (MS). Dr. A. Kumar is Acting Chief of MSM, and Dr. G. Williamson is Chief of MS.

COL M. D. Remus is Commander and Director of CERL, and Dr. L. R. Shaffer is Deputy Director.



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FAILURE ANALYSIS OF OZARK, ARKANSAS, POWER PLANT SOCKET-HEAD CAP SCREWS

1 INTRODUCTION

Background

The hydroelectric power generating system at the Ozark, AR facility is unique, because it is one of two plants in the world whose turbines are positioned horizontally rather than vertically. The Ozark Power Plant, which is operated by the Little Rock District of the U.S. Army Corps of Engineers, has five turbines and generators that are remotely controlled and monitored at the Dardanelle (AR) Power Plant. The turbine blades span approximately 32 ft (9.75 m) and operate with a normal 32-ft (9.75 m) head. During operation, the turbines revolve at 60.24 rpm; gearboxes located between the turbines and generators increase the speed to 514 rpm for operating the generators.

The turbine blades are connected to a housing containing the mechanism that controls blade pitch. This housing is part of the turbine shaft assembly, which is joined to the shaft by 36 socket-head cap screws. The cap screws, manufactured by Allis-Chalmers Corporation, are fabricated from American Iron and Steel Institute (AISI) 4340 low-alloy steel and are specified to be heat-treated, i.e., austenitized, quenched, and tempered to a Rockwell C (R_C) hardness of 38 to 45.

After approximately 10,000 hr (4×10^7 cycles) of operation, fractured cap screws were found in turbine units 3 and 4. Researchers learned that when the turbines were installed, the cap screws were tightened into place by using a sledgehammer and a 3-ft-long (91.5 cm) hex key; the desired preload on these cap screws was 15 ksi (103 MPa).

Prior to the profusion of cap screw failures, i.e., after 1000 hr of operation, several cap screws were sent to the Allis-Chalmers Advanced Technology Center, Milwaukee, WI, for stress corrosion analysis. Their tests showed that the cap screws were immune to stress corrosion cracking in the Arkansas River water. An analysis conducted at the U.S. Army Construction Engineering Research Laboratory (CERL) after numerous cap screw failures also found no evidence of stress corrosion cracking.

When the cap screws began failing, Allis-Chalmers designed a stud bolt and nut arrangement to replace the socket head cap screws. Little Rock District purchased the new bolts and nuts and installed them in turbines 3 and 4. This installation included spot-facing the nut seats to insure proper alignment of the stud bolts.

Objective

The objectives of this investigation were: (1) to determine the cause of socket-head cap screw failures in the turbines of the Ozark Power Plant facility; and (2) to analyze the redesigned stud and nut arrangement currently used to replace the socket-head cap screws.

Approach

A systematic procedure was used to ascertain the cause of the socket-head cap screw failures. Four failed cap screw specimens were first visually examined at CERL for evidence of corrosion pitting, uniform corrosion, and surface cracks, followed by a dye penetrant inspection for surface cracks and related defects. Next, the cap screws were sectioned and machined, and a hardness survey was conducted over the cross sections. Additional sections were cut for microstructure analysis and microscopic examination of the screw threads. The fracture surfaces were cleaned and examined optically at low magnification. Selected portions were examined at low and high magnifications in the CERL scanning electron microscope (SEM). Several cap screw sections were selectively heat-treated and checked for hardness. Finally, a mechanics analysis of the cap screw geometry and loading was used to predict the cause and number of cycles (or hours of operation) on cap screws heat-treated to different hardnesses.

2 MATERIALS AND SPECIFICATIONS

The federal specification which governed the purchase of the alloy-steel cylindrical-head cap screws was FF-S-86b. The latest revision of this specification, FF-S-86d,¹ dated 15 June 1971, is basically the same as FF-S-86b, having the following salient requirements:

- 3.2.1 (a) Ultimate tensile strength (S_u), sizes greater than
0.190 in. (0.483 cm)
170 ksi (1172 MPa)

¹ Screw, Cap, Socket-head, Federal Specification FF-S-86d (General Services Administration, 1971).

(b) Rockwell C hardness (R_C) (after heat treatment)
... 38-45 R_C

3.5.4.1 *Method of Manufacture (of screw threads)* - Unless otherwise stated, the method employed for production of screw threads on cap screws shall be at the option of the manufacturer.

3.6.1 *Carburization and Decarburization* - There shall be no evidence of carburization or total decarburization on the thread surfaces.

4.1 *Responsibility for Inspection* - Unless otherwise specified in the contract or purchase order, the supplier is responsible for the performance of all inspection requirements as specified ...

Additional manufacturing requirements are referred to in American Society for Testing and Materials (ASTM) Standard Specification A 574-67.² Some of the more specific requirements of ASTM are:

3.2 Unless otherwise specified, the heads of screws ... over 1.500 in. (3.81 cm) diameter ... may be fabricated by hot or cold forging or by machining. Sockets may be forged or machined.

3.3 ... For diameters greater than 0.625 in. (1.59 cm), threads may be rolled, cut, or ground.

3.4 The screws shall be heat treated by oil quenching from above the transformation temperature and then tempered at a temperature not lower than 650°F (345°C).

5.1 The hardness of finished screws shall be ... 37 to 45 R_C for [those] 0.250 in. (0.64 cm) and larger.

The material selected by Allis-Chalmers for the socket-head cap screws used in the turbines of the Ozark Power Plant was AISI 4340 steel* (Table 1 indicates the composition of the steel); however, spectrographic analysis of a sample cap screw showed that it was fabricated from AISI 4140 alloy steel. The cap screws purchased for use in the Ozark Power Plant had a black oxide coating.

3 RESULTS AND DISCUSSION

Visual Inspection

Four failed socket-head cap screw specimens were studied; three of the specimens were sent to CERL

² *Alloy Steel Socket Head Cap Screws*, ASTM Standard Specification A 574-67 (American Society for Testing and Materials, 1967).

*AISI 4340 alloy steel was not specifically required by Federal Specification FF-S-86b. However, AISI 4340 was mentioned in a letter sent to CERL 26 September 1975 by J. W. Marlow and in Allis-Chalmers Advanced Technology Center Investigation No. 3517, January 1975.

Table 1
Chemical Analysis of AISI 4340 Steel

Element	Specified Weight, %	Analysis Weight, %
C	0.38 to 0.43	0.39
Mn	0.60 to 0.80	0.84
P	0.035 Max	0.005
S	0.040 Max	0.018
Si	0.20 to 0.35	0.22
Ni	1.65 to 2.00	0.01
Cr	0.70 to 0.90	0.89
Mo	0.20 to 0.30	0.21
V	—	0.01

(Figures 1, 2, and 3), and researchers obtained the fourth during a later site visit (Figure 4). The three specimens sent to CERL had failed at the fillet where the head and shank joined. When they were examined, it was found that the fracture surfaces were deeply corroded and that the topography of the fracture was completely obliterated. The corrosion scale and surface oxide coating were removed chemically and the cap screws examined again. Although removing the oxide had eliminated the microscopic fracture details, the macroscopic fracture surface still remained.

Examination of the fracture in cap screw #25 (Figure 1) revealed what appeared to be a high-cycle fatigue marking (smooth area). Adjacent to this was a much coarser region which may have been the result of low-cycle fatigue (large plastic strains ahead of the crack tip). Final fracture occurred when the surviving ligament broke (the very coarse inclined region). The size of the uncracked ligament indicates that the alternating loads were relatively low (approximately 20 to 40 ksi [138 to 276 MPa]). Similar observations were made about the other failed cap screws; however, as shown by the size of the last ligament that failed, cap screw #31 (Figure 3) failed entirely in fatigue due to the absence of the remaining ligament. This was probably one of the first cap screws to fail and was the one having the lowest alternating load. Cap screw #23 (Figure 2) failed at a higher load, probably at some time between the failures of cap screws #31 and #25. The fracture of the cap screw brought from the power plant (cap screw OZ, Figure 4) occurred in the threads shortly before the turbine was shut down for servicing. There was very little corrosion either on the fracture surface or anywhere else along the cap

screw. Examination of this fracture definitely indicated high-cycle fatigue, and close inspection revealed multiple sources of crack initiation. The large coarse region among the small fatigued regions indicated that the final fracture occurred at loads much higher than those causing the failure of the other cap screws examined.

Examination of the threaded portions of the failed cap screws showed extensive corrosion, with severe pitting in some of the threads. The corrosion may or may not have reduced the fatigue initiation life of the cap screws; however, it probably rapidly accelerated the propagation of the fatigue crack.

Dye Penetrant Inspection

Two of the cap screws were inspected by means of a dye penetrant (Figure 5). The surfaces were first thoroughly cleansed, sprayed with penetrant, wiped clean, and sprayed with developer. This procedure was repeated several times. No cracks or other surface defects were found. The results indicate that fracture resulted from fatigue in the most critical location, i.e., the cap screw fillet. The absence of cracks elsewhere along the cap screw means that the threads are not as severe a stress concentrator as the fillet.

Hardness Surveys

R_C hardness measurements were made on cross sections cut from each of the four cap screw specimens. Figures 6, 7, 8, and 9 show results of the hardness surveys. The hardness of cap screw OZ (Figure 9) ranged from 15 to 28 R_C with an average of approximately 20 R_C. This is much below the specified hardness range of 38 to 45 R_C. Similarly, cap screw #23 (Figure 6) had a hardness range of 24 to 27.5 R_C, which was slightly higher and more uniform than cap screw OZ, but still much below specifications. Cap screw #25 (Figure 7) had a hardness range of 22.5 to 29.5 R_C, with the hardest regions lying near the outer surface. Cap screw #31 (Figure 8) ranged in hardness from 28 to 32 R_C.

Results of the hardness surveys show that in each case the cap screw was too soft; i.e., hardness was less than the 38 to 45 R_C specified. The hardness over the cross section was generally uniform, with the threaded surfaces being somewhat harder than the centers. The low hardness readings indicate that the cap screws were improperly heat-treated and that manufacturer quality control was lacking.

Metallography

Sections were taken from the cap screws and prepared for microstructure examination by grinding, polishing, and etching with 2 percent nital solution. The macrostructure of the polished and etched specimens showed a coarsely banded structure (Figure 10) caused either by massive segregation of the alloying elements or by plastic flow associated with a forging operation. Examination of the microstructure of the as-received sample revealed that the coarse banding was composed of alternating regions of coarse martensite and acicular ferrite. Figures 11a and 11b show the microstructure at magnifications of 200x and 500x, respectively. The light, needle-shaped regions are ferrite (α iron) and the uniformly gray regions are martensite; the darker elongated particles are nonmetallic inclusions common to most steels. The presence of ferrite indicates an improper heat treatment, since the microstructure should be entirely tempered martensite.

High-magnification inspection showed that the threads were machined rather than rolled. No decarburization of the threads was observed.

Heat Treatment Simulation

As a result of the microstructure examinations and the low measured hardnesses of the cap screws in the as-received condition, it was decided to conduct a series of heat treatments aimed at obtaining the desired microstructure and hardness, and to determine the cause of the improper microstructures and associated low hardnesses.

To obtain the desired microstructure, cap screw sections were normalized at 1600°F (870°C) for 1 hr and air-cooled. The normalization heat treatment, which served to improve the alloy's homogenization and consequently reduce segregation banding, was followed with a 1550°F (843°C) austenization heat treatment for 1 hr and then oil-quenching. (Austenitizing is the heat treatment used to dissolve all the carbon into solution in order to optimize an alloy's hardness and mechanical properties.) The as-quenched hardness ranged from 48 to 55 R_C. The specimen was then tempered 3½ hr at 800°F (423°C), with the resultant hardness ranging from 39 to 41 R_C. The microstructure was entirely tempered martensite; note the absence of ferrite, as shown in Figures 12a and 12b at 200x and 500x, respectively.

Obtaining the microstructure seen in the as-received cap screws required more effort. There are

two probable causes of the banded, acicular ferrite (1) the temperature of austenitization before quenching was too low, or (2) the cooling rate was too slow to properly quench the steel. The latter was believed to be the more plausible cause, because the ferrite in the microstructure was acicular rather than equiaxed or spherical, and because it tended to outline the prior austenite grain boundaries in regions where the martensite transformation occurred.

Experimental heat treatments showed that this was indeed true, since specimens austenitized below the required heat-treating temperature contained more equiaxed ferrite, and because the hardness was approximately the same as that of specimens which were properly heat-treated. The microstructure seen in the as-received cap screws was obtained when specimens were heat-treated at the proper austenitizing temperature, but quenched at different cooling rates. This process was accomplished by selecting a long, narrow specimen and placing it on a steel heat sink to induce a cooling rate gradient along its length. The portion closest to the steel heat sink cooled very rapidly, whereas the portion at the outer end exposed to the air cooled slowly. The desired microstructure was found at about two-thirds of the distance from the heat sink (Figure 13). The hardness of this region ranged from 27 to 34 R_C before tempering and 21 to 26 R_C after tempering, indicating that the cap screws which failed in service were improperly heat-treated, with one of the following conditions being the cause:

1. There was no heat treatment after machining, i.e., the cap screws were left in the annealed condition.
2. The quenchant was too hot.
3. Too much time was taken in transferring the cap screws from the furnace to the quench.

High Magnification Examination of the Fracture Surface

The fractured cap screw brought from the Ozark Power Plant was relatively uncorroded. As a result, the fracture surface still retained most of the topology it had when it first fractured. The fracture surface was lightly cleaned and then examined in the SEM at various magnifications. Figure 14 shows a region where a fatigue crack initiated and became larger. The smooth region on the upper left is a result of high-cycle fatigue crack propagation; the

coarse region adjacent to it is low-cycle fatigue which occurred near the end of the cap screw's life. Examination of the screw threads showed machining marks. Figure 15 shows a region separating high-cycle fatigue (left region) from rapid fracture (ridged region).

Mechanics Analysis of Cap Screw Failure

Most of the cap screw failures occurred at the fillet joining the shank to the head. A stress analysis of the socket-head cap screw was made based on its similarity to a T-head component under load. Using the following cap screw dimensions, the value of the stress concentration of the fillet could be determined:

$$D = 4.5 \text{ in. (11.43 cm) head diameter}$$

$$d = 3.125 \text{ in. (7.94 cm) shaft diameter}$$

$$r = 0.125 \text{ in. (0.32 cm) fillet radius}$$

$$m = 3.0 \text{ in. (7.62 cm) head height}$$

$$l = 0.313 \text{ in. (0.79 cm) distance from shank to mid-contact point on head}$$

$$D/d = 1.44$$

$$r/d = 0.04$$

$$\frac{ld}{m^2} = 0.109.$$

For this particular cap screw, the stress concentration (K_t) was found to range from 5 to 7.³ Once the static stress concentration factor was known, it was possible to calculate the value of the fatigue strength reduction factor (K_f), which is necessary to predict the life of a notched member. The equation for K_f is:

$$K_f = 1 + \frac{K_t - 1}{1 + a/r} \quad [\text{Eq 1}]$$

where r = notch root radius
 a = a material constant given by

$$a = 10^{-3} \left(\frac{600}{\text{BHN}} \right)^{1.8} \quad [\text{Eq 2}]$$

where BHN = Brinell hardness number.

³R. E. Peterson, *Stress Concentration Factors* (John Wiley and Sons, Inc., 1974), pp 255-256; 275; 280.

Values of K_f were calculated for the cap screw, assuming that $K_t = 3, 5$, and 7 , and assuming a hardness of 245 and 350 BHN (for 23 and 38 R_c , respectively). Table 2 gives the calculated K_f values. Knowing these values, fatigue crack initiation life can then be calculated by one of two methods. The first method uses a stress vs. cycles-to-failure curve for the material of interest; the second method is based on the notch root plasticity and requires strain vs. cycles-to-failure data. Stress vs. cyclic life data are available for use in the first method.⁴ Using a K_t of 3.3 , a mean stress of 25 ksi (172 MPa), and an alternating stress of ± 10 ksi (69 MPa), the predicted life is infinite. It would be reduced considerably for a K_t greater than 3.3 . However, at an alternating stress of ± 18 ksi (124 MPa), the predicted life is 10^6 to 10^7 cycles.

Table 2
Calculated K_f Values

BHN	a	K_t	K_f
245	0.005	3	2.9
		5	4.8
		7	6.8
350	0.003	3	2.9
		5	4.9
		7	6.9

The second method is more accurate in this instance, because the high K_t values cause large-scale plasticity to occur at the notch root, i.e., the fillet. However, the calculations are much more involved.⁵ The notch root behavior can be quantified using Neuber's equation:⁶

$$K_f (\Delta \sigma \Delta \epsilon E)^{1/2} = (\Delta \sigma \Delta \epsilon E)^{1/2} \quad [\text{Eq 3}]$$

where $\Delta \sigma$ = the stress range in the gross section
 $\Delta \epsilon$ = the strain range in gross section
 E = Young's modulus
 $\Delta \sigma$ = the true stress at the notch root
 $\Delta \epsilon$ = the true strain at the notch root.

In this instance, the cyclic loads are assumed to be very small, so that the gross section behaves elastically; hence

$$\Delta \sigma = E \Delta \epsilon \quad [\text{Eq 4}]$$

⁴ *Structural Alloys Handbook*, Vol 1 (Mechanical Properties Data Center, Belfour Stulen, Inc., 1976).

⁵ B. I. Sandor, *Fundamentals of Cyclic Stress and Strain* (University of Wisconsin Press, 1972), pp 113-140.

⁶ Sandor, pp 113-140.

Similarly, the true elastic strain at the notch root (ϵ_e) is

$$\epsilon_e = \frac{\Delta \epsilon_e}{2} = \frac{\sigma}{E} = \frac{\Delta \sigma}{2E} \quad [\text{Eq 5}]$$

Combining Eqs 3, 4, and 5 yields

$$K_f (\Delta \sigma \frac{\Delta \sigma}{E} E)^{1/2} = (\Delta \sigma \Delta \epsilon E)^{1/2} \quad [\text{Eq 6}]$$

$$K_f \Delta \sigma = (\Delta \sigma \Delta \epsilon E)^{1/2}$$

$$= 2 \left(\frac{\Delta \sigma}{2} \frac{\Delta \epsilon}{2} E \right)^{1/2}$$

$$= 2 (\sigma \epsilon_e E)^{1/2}$$

$$= 2 (E \epsilon_e \epsilon_t E)^{1/2}$$

$$K_f \Delta \sigma = 2 E (\epsilon_e \epsilon_t)^{1/2}$$

where ϵ_t = the total true strain at the notch root.

The alternating stress is then given by

$$\frac{\Delta \sigma}{2} = \frac{E (\epsilon_e \epsilon_t)^{1/2}}{K_f} \quad [\text{Eq 7}]$$

Strain vs. cyclic life data for AISI 4340 steel were found for the hardness of interest in the *Ford Motor Company Databook*.⁷ The strain vs. life curves were modified to account for a mean stress of 25 ksi (172 MPa); these curves are presented in Figure 16 for a hardness of 23 R_c (243 BHN) and in Figure 17 for a hardness of 38 R_c (350 BHN). Using these data, a Neuber analysis of the notch equation (Eq 7) can be used to compile a stress vs. cyclic life curve for a notched member (Figure 18).

Using this curve, if the cap screw is preloaded to 15 ksi (103 MPa) and then cyclically loaded to 35 ksi (241 MPa) and back to 15 ksi (103 MPa) (i.e., it has mean stress of 25 ksi [172 MPa] and alternating stress of 10 ksi [69 MPa]), the life would definitely depend on the hardness and the value of K_f . That is, for a hardness of 23 R_c (ultimate tensile strength $S_u = 105$ ksi [723 MPa]) and a K_f value of 7 , the predicted crack initiation life would be 6×10^4 cycles (2 hr of operation); if the K_f was 5 , then the predicted life would be approximately 10^8 cycles (3150 hr). The

⁷ "Monotonic and Cyclic Properties of Engineering Materials," *Ford Motor Company Databook* (undated publication).

best demonstration of how much the proper heat treatment would increase the life can be seen in the difference between the curves for different hardnesses. At a hardness of 38 to 40 R_C ($S_u = 180$ ksi [1240 MPa]), the crack initiation life for a K_f value of 7 would be approximately 10^7 cycles (300 hr); for K_f values of 5 and 3, the predicted lives are virtually infinite.

If the mean stress is neglected, a similar curve is obtained, but the predicted lives are longer. The mean stress is particularly deleterious for low-strength, severely notched members such as a low-hardness cap screw. Calculations made assuming the fillet radius was doubled showed little improvement in predicted fatigue life. The analysis showed that the best way to obtain optimum life was to properly heat-treat the cap screws to provide sufficient strength for fatigue resistance, while keeping the hardness low enough to prevent stress corrosion cracking problems.

Fatigue Assessment of Cap Screw Replacement Stud Bolt

A simple analysis was performed on the threaded stud and nut arrangement designed to replace the socket-head cap screws now in use. This arrangement is expected to have a greater fatigue life than the cap screws, mainly because the severe stress concentration present in the fillet joining the head and shank of the cap screw is eliminated. The greatest stress concentration in the stud and nut arrangement occurs at the first two threads where the nut and stud meet. Sometimes the transmission of load through the mating thread faces induces a bending stress at the thread roots, which acts in conjunction with the axial loads to decrease the life of the nut and stud assembly. Work to date has indicated that the stresses in the first engaged thread of a nut and bolt can be as high as four times the mean load.⁸ The fatigue life of a nut and bolt can be increased if the nut is carefully machined to insure uniform contact along its length. Furthermore, it has been shown that increasing the preload imposed on a nut and bolt generally tends to increase the fatigue life.

The values of the stress concentration factor for a nut and bolt arrangement have been reported to range from 2.7 to 6.7, depending on the source.⁹ A

⁸ N. E. Frost, K. J. Marsh, and L. P. Pook, *Metal Fatigue* (Oxford University Press, 1974), pp 401-408.

⁹ R. E. Peterson, *Stress Concentration Factors* (John Wiley and Sons, Inc., 1974), pp 253-254.

K_t of 4 is probably most realistic. For the stud design to replace the cap screw, the K_t will probably be low, due to the reduced shank diameter between the threads. The fatigue life of this design can be further improved if the threads are rolled on rather than machined after heat-treating. This will also increase the resistance to stress corrosion cracking. Every shipment of studs received should be checked to insure that they have been properly heat-treated.

4 CONCLUSIONS

The following conclusions are based on research conducted on the failure of Ozark socket-head cap screws.

1. The failure of the cap screws was caused by fatigue.
2. The fatigue strength of the cap screws was reduced by improper heat treatment. The results of the hardness survey conducted on each cap screw showed that in each case, the hardness was less than the 38 to 45 R_C specified in FF-S-86b.
3. The socket-head cap screws generally failed at the fillet joining the shank and the head. The stress concentration (K_t) of the 0.125-in. (.322 cm) radius fillet ranged between 5 and 7, and resulted in a numerically similar fatigue strength reduction factor, K_f .
4. The failure was not caused by stress corrosion cracking; however, corrosion fatigue caused increased crack propagation and, consequently, reduced life.
5. Proper heat treatment to a hardness of 38 to 40 R_C will enhance a cap screw's high-cycle fatigue life. Hardness greater than 44 R_C should be avoided to prevent the problem of stress corrosion cracking.
6. The stud and nut arrangement presently being used to replace the socket-head cap screws appears to have better fatigue life than the cap screws. This is based on a decrease in the fatigue strength reduction factor (K_f) due to the absence of a fillet and the stud geometry.

5 RECOMMENDATIONS

The following procedures are recommended to insure that future cap screw and stud/nut failures

are minimized:

1. Insure that all cap screws or studs are heat-treated in the following manner:

a. Provide full normalizing heat treatment (after forging) at 1600°F (870°C).

b. Austenitize for 2 to 3 hr at 1550°F (843°C), and quickly oil quench.

c. Temper at 800°F (423°C) to a hardness of 38 to 42 R_C (approximately 3½ hr).

2. Use rolled rather than machined threads for both cap screws and studs. The threads should be rolled on after heat treatment.

3. Keep cap screw and stud hardness below 44 R_C to prevent stress corrosion cracking.

4. Machine cap screw and stud/nut seats to insure good alignment and perpendicularity; this will eliminate bending stresses.

5. In the future, keep the threaded portion of cap screws not closer than 1 in. (2.54 cm) to the head. Similarly, the fillet should be increased to a minimum of 0.5 in. (1.27 cm) radius and locally shot-peened.

6. The preload used when installing the cap

screws or stud/nut arrangement should be between 15 and 20 ksi (117 to 138 MPa) and should be imposed equally on all cap screws or stud/nut arrangements used in the turbine.

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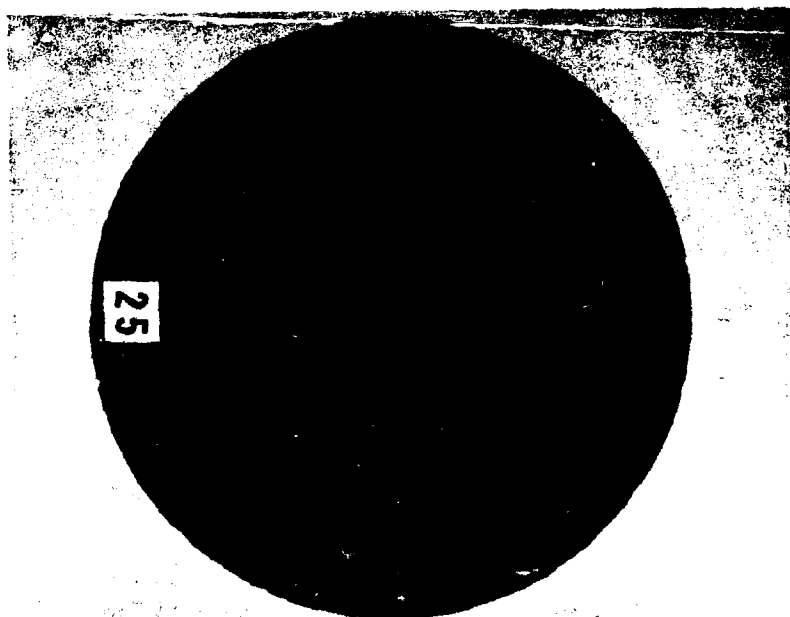
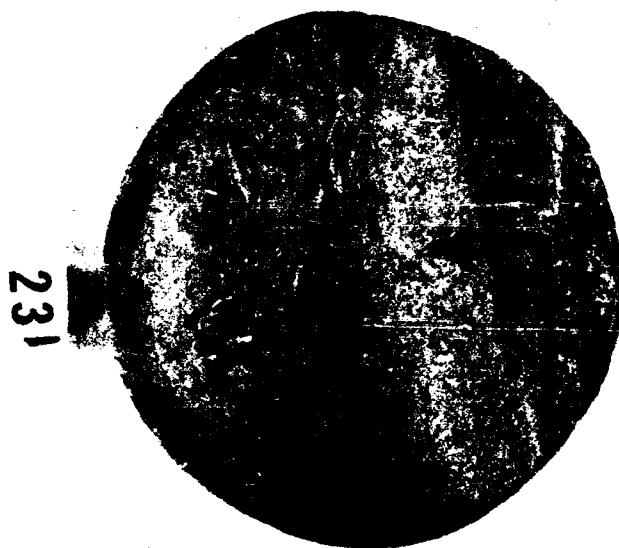


Figure 1. Fracture surface of cap screw head #25 showing heavily corroded surface, 0.8x.



(a) Cap screw shaft, 1x.



(b) Cap screw head, 0.7x.

Figure 2. Fracture surface of cap screw shaft (a) and head (b) #23. Fatigue markings can be seen in the center of the fracture surfaces.

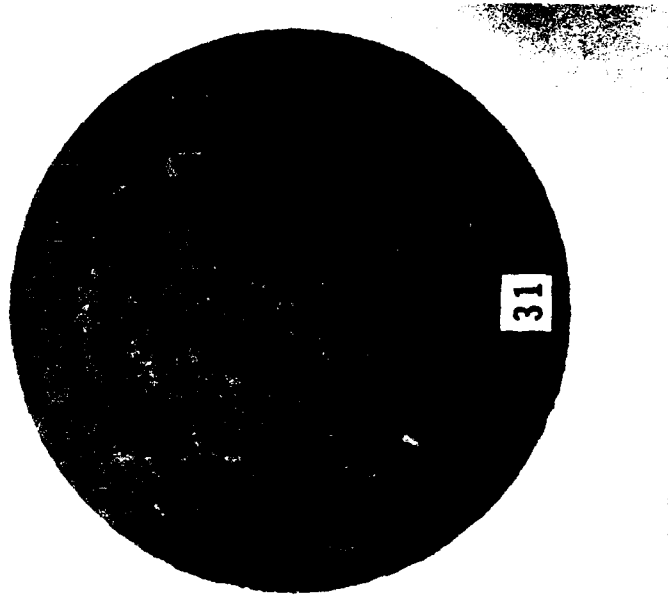


Figure 3. Fracture surface of cap screw head #31 showing heavily corroded surface, 0.7x.

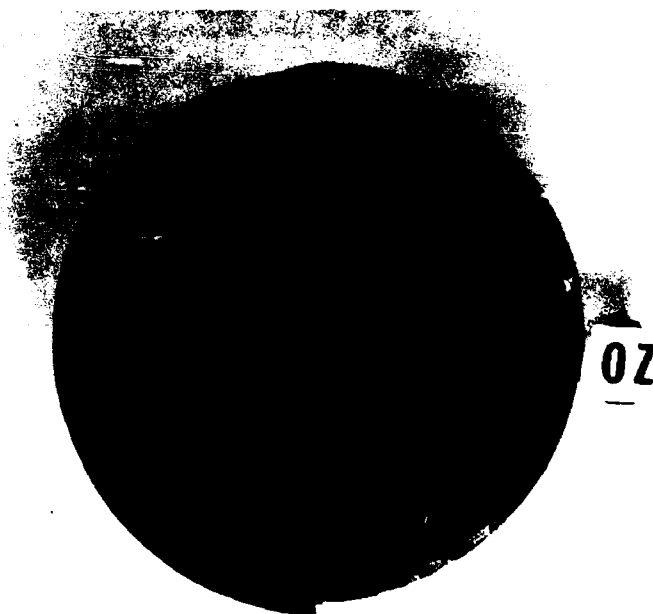
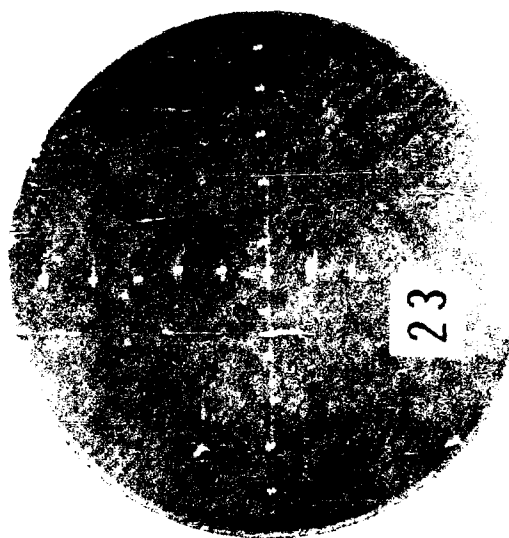


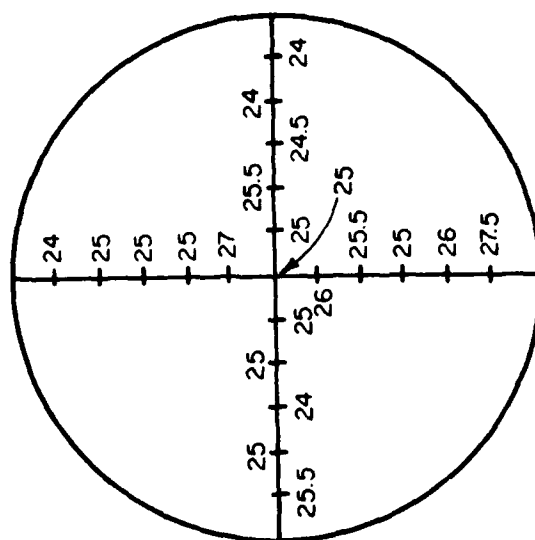
Figure 4. Fracture surface of cap screw OZ, 1x. Failure occurred in the threads after only a small amount of fatigue crack propagation. The smooth region on the left is the extent of the fatigue crack propagation until fracture. Note the presence of other fatigue crack initiation sites located along the threaded outer surface.



Figure 5. Dye penetrant examination of two cap screws, 0.3x. No cracks were detected on the outer surfaces.



(a)

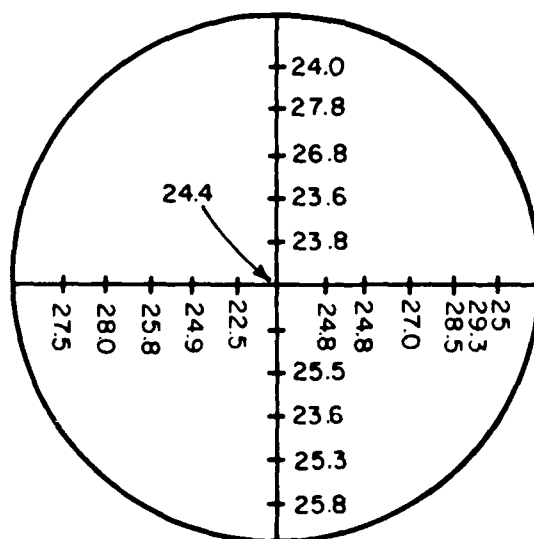


(b)

Figure 6. Rockwell C hardness survey on cap screw #23 (a) and the numerical values (b), 1x.

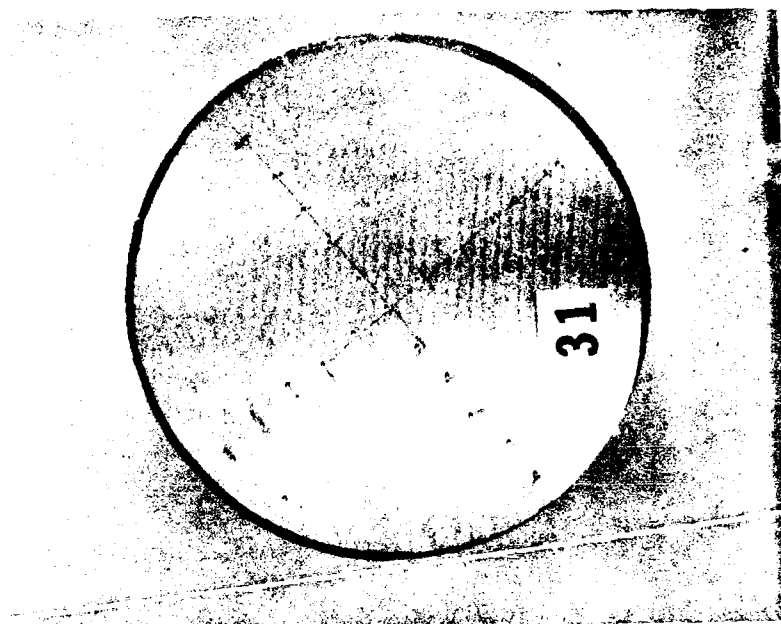


(a)

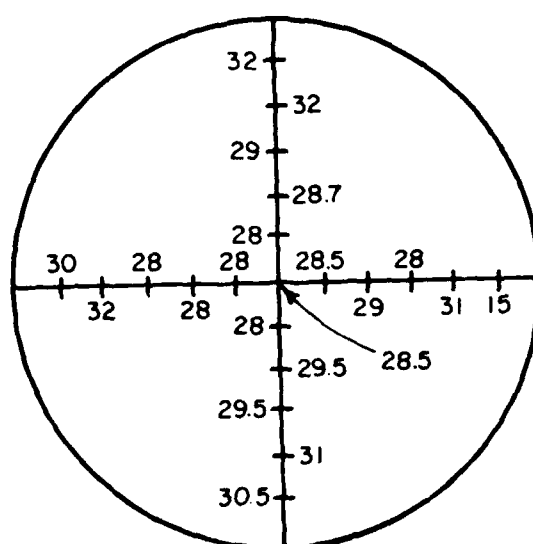


(b)

Figure 7. Rockwell C hardness survey on cap screw #25 (a) and the numerical values (b), 1x.

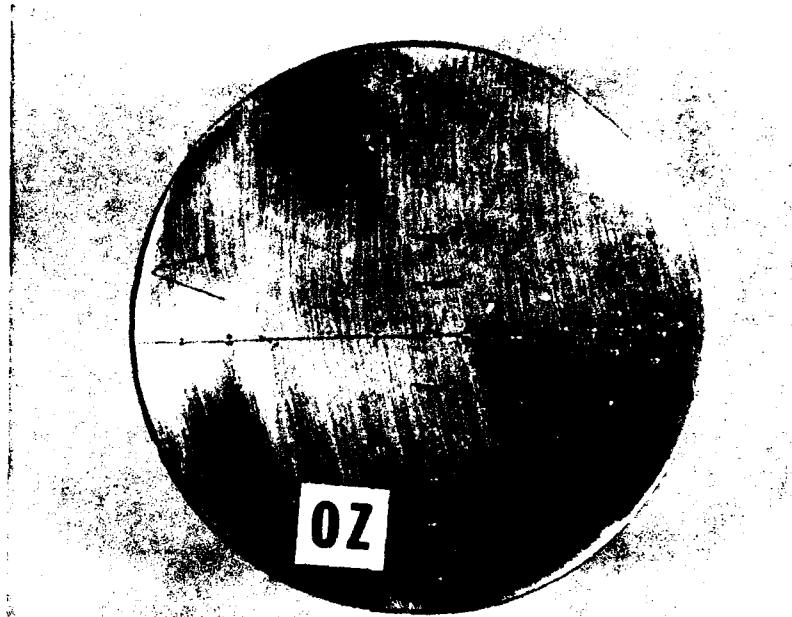


(a)

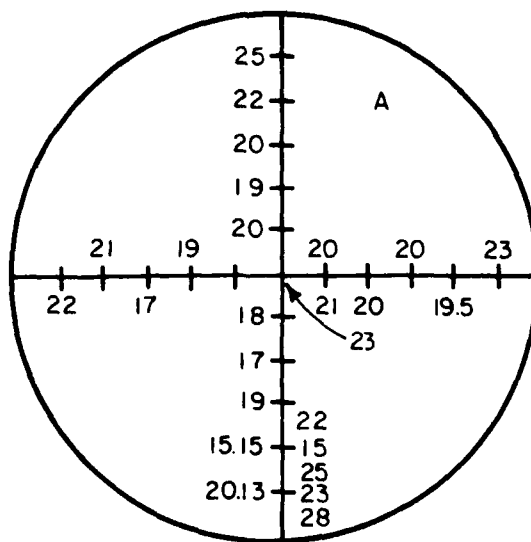


(b)

Figure 8. Rockwell C hardness survey on cap screw #31 (a) and the numerical values (b), 1x.



(a)



(b)

Figure 9. Rockwell C hardness survey on cap screw OZ (a) and the numerical values (b), 1x.

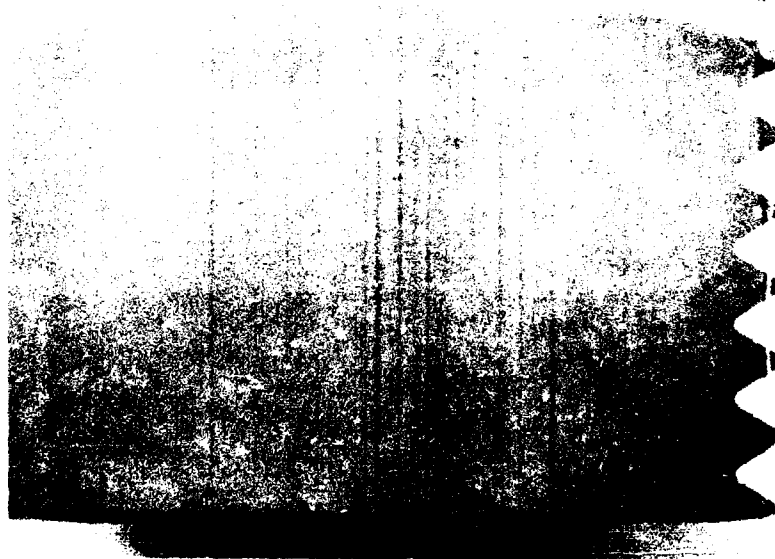
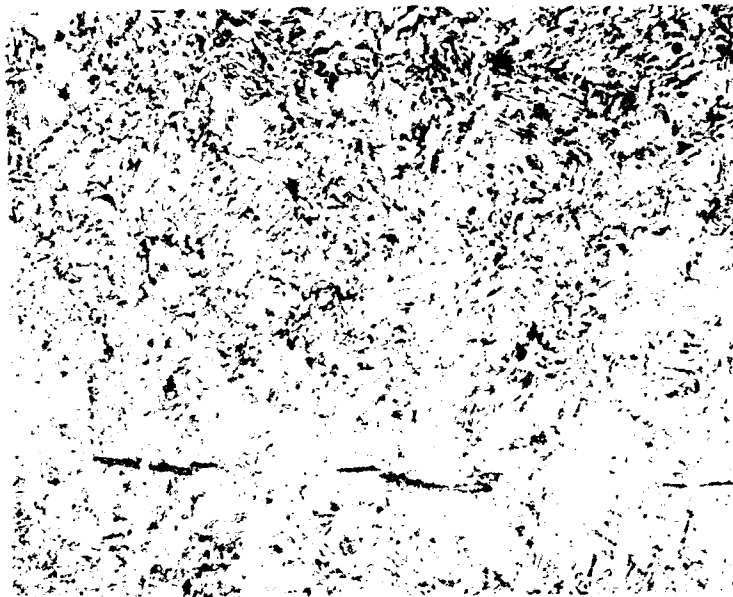
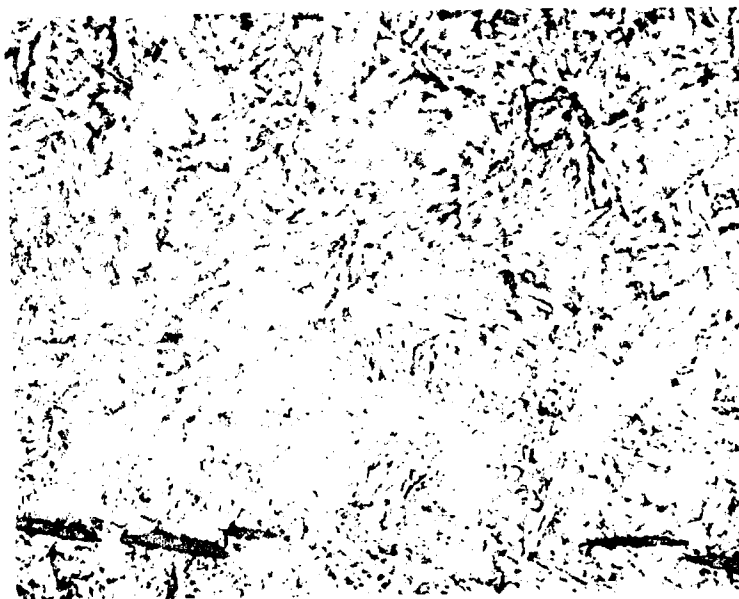


Figure 10. Macrograph of a cap screw section showing flow lines and/or severe segregation banding, 3.3x.

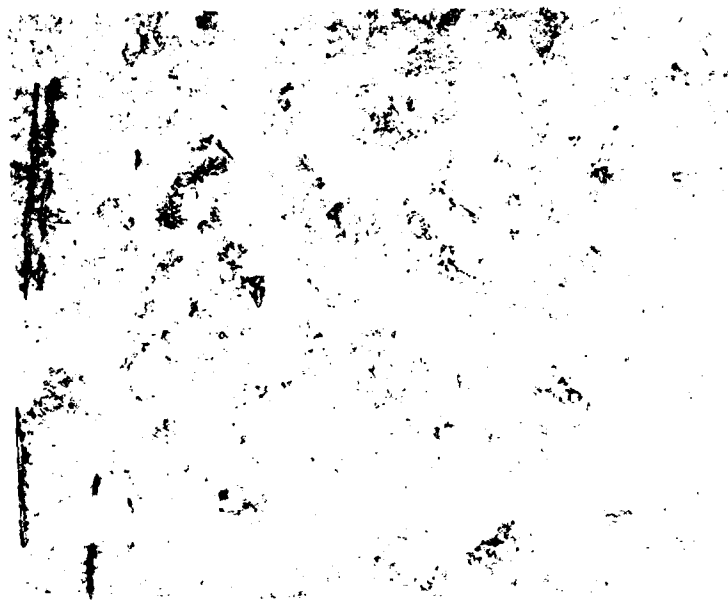


(a) 200x.

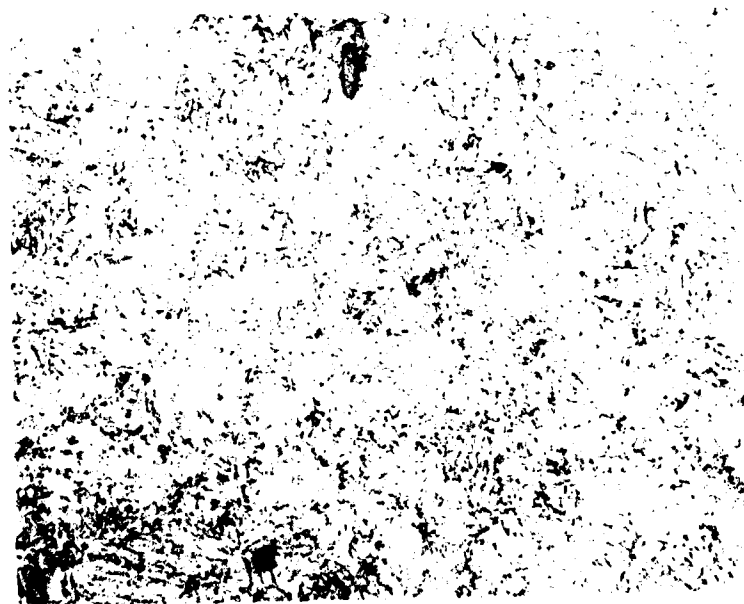


(b) 500x.

Figure 11. Microstructure of as-received cap screw. Note the acicular ferrite and the martensite region (uniform color).



(a) 200x.

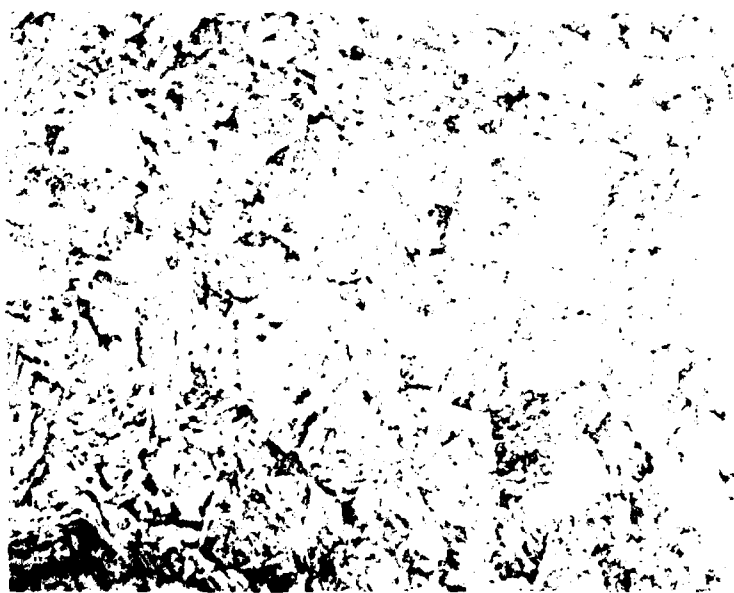


(b) 500x.

Figure 12. Microstructure of properly heat-treated cap screw section. The microstructure is entirely tempered martensite and is much less coarse than the martensite in the as-received condition. The elongated dark gray regions are nonmetallic inclusions commonly found in steels.



(a) 200x.



(b) 500x.

Figure 13. Microstructure of cap screw sections heat-treated to obtain the microstructure of the as-received cap screw.



Figure 14. SEM micrograph of the OZ cap screw fracture, 10x. Note regions of fatigue crack initiation and propagation. Coarse machining marks can be observed on the thread area.

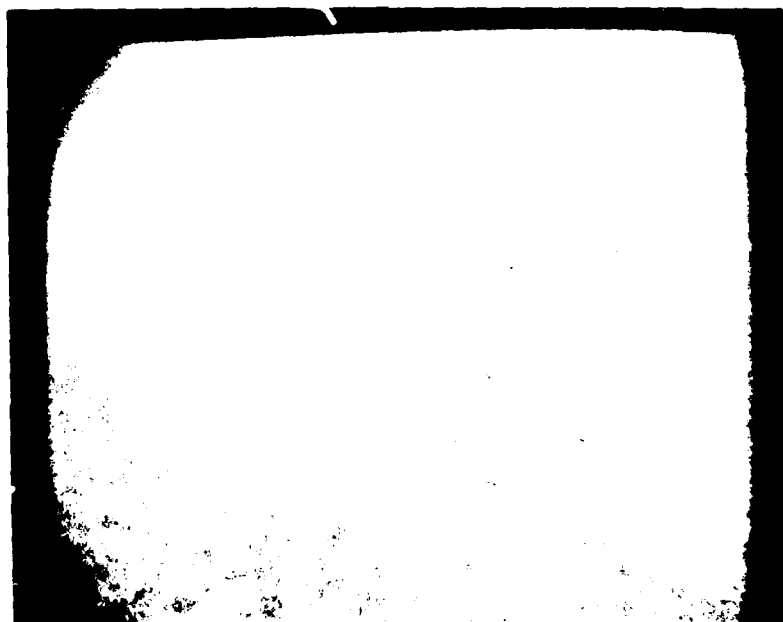


Figure 15. SEM micrograph showing transition from fatigue crack growth (left) to rapid fracture (right), 18x.

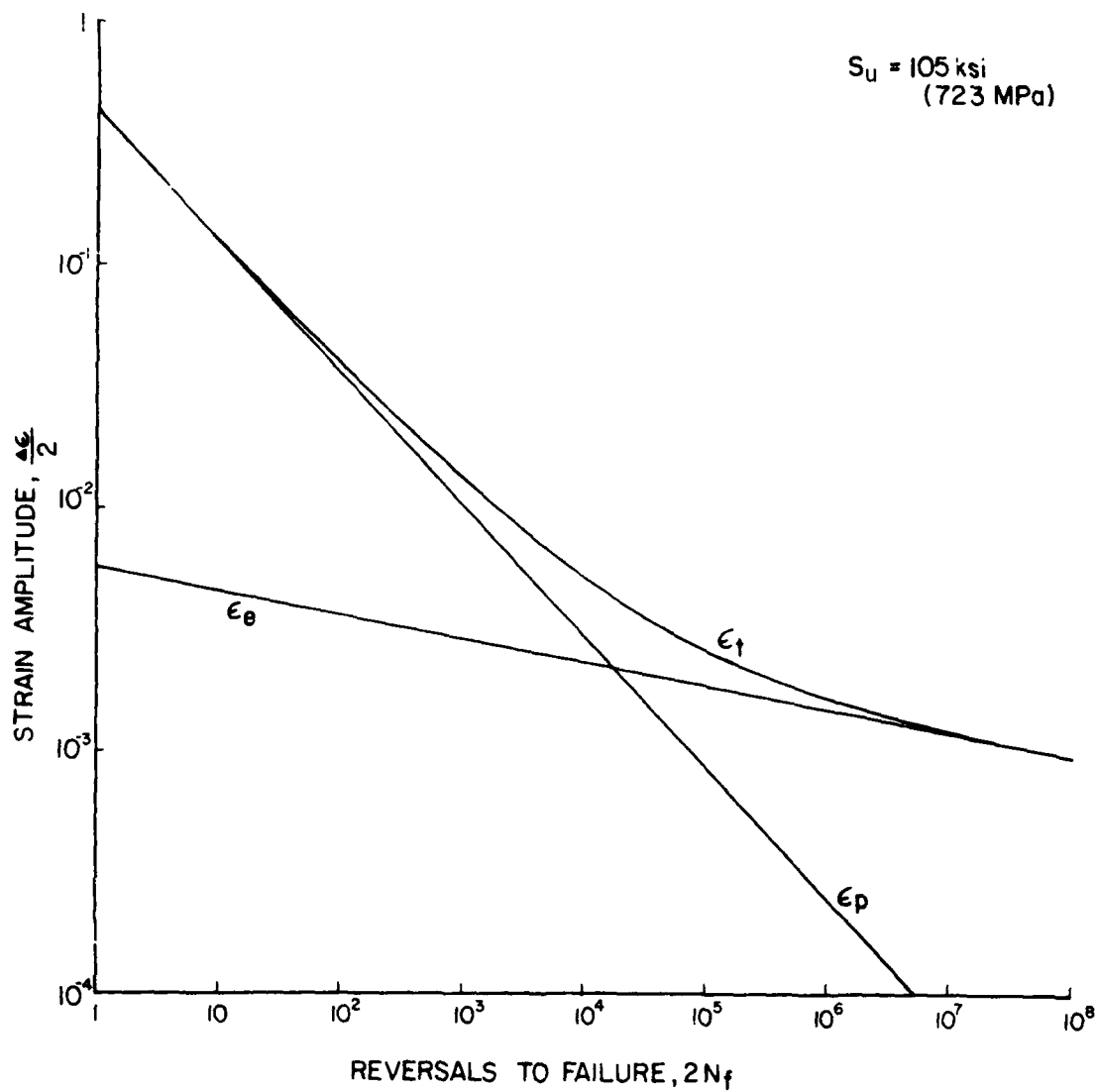


Figure 16. Strain amplitude vs. fatigue crack initiation life for AISI 4340 steel heat-treated to hardness of 23 R_C and having a 25-ksi (172 MPa) mean stress. (The above data are for smooth specimens.)

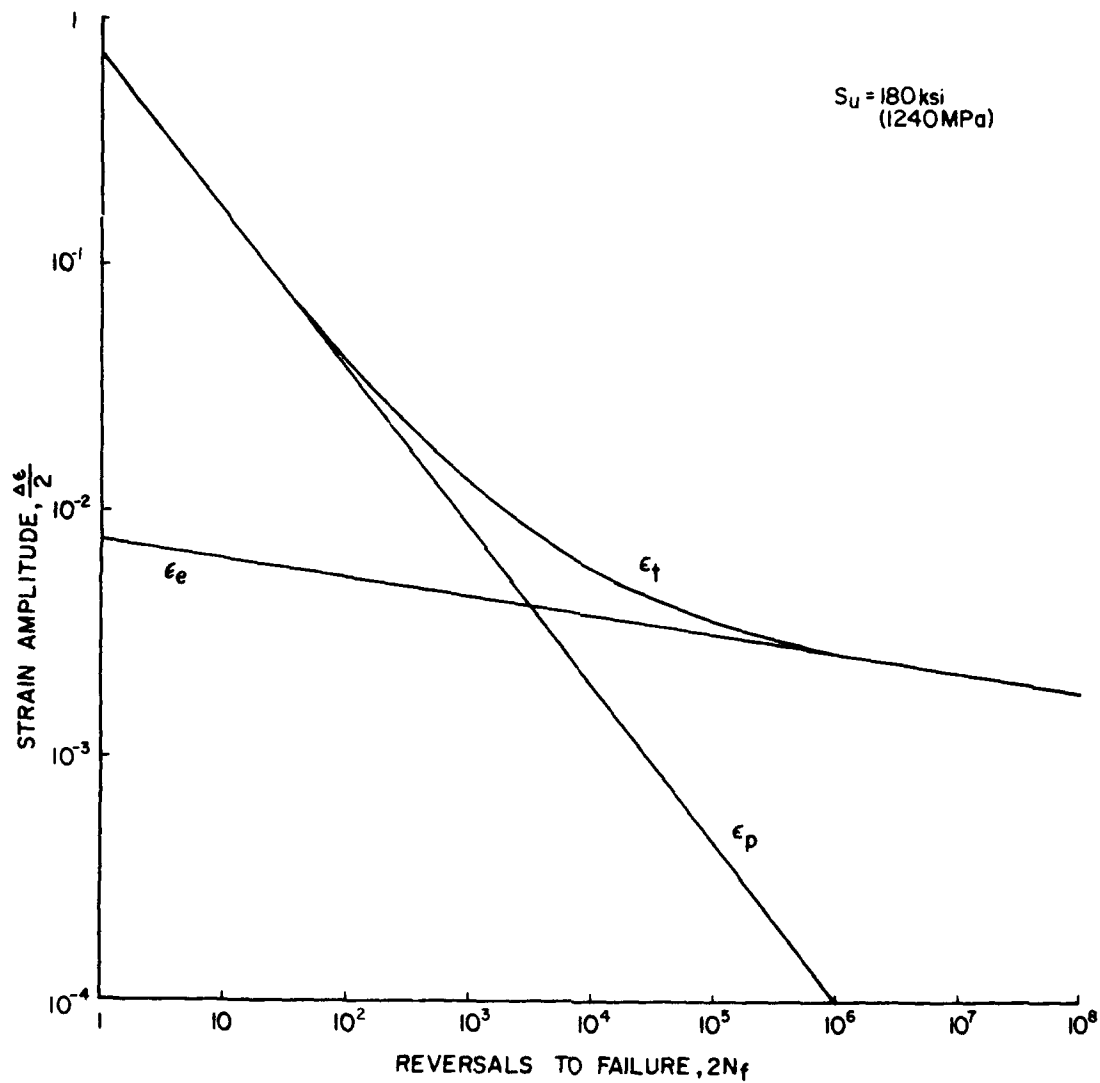


Figure 17. Strain amplitude vs. fatigue crack initiation life for AISI 4340 steel heat-treated to hardness of 38 R_C and having a 25 ksi (172 MPa) mean stress. (The above data are for smooth specimens.)

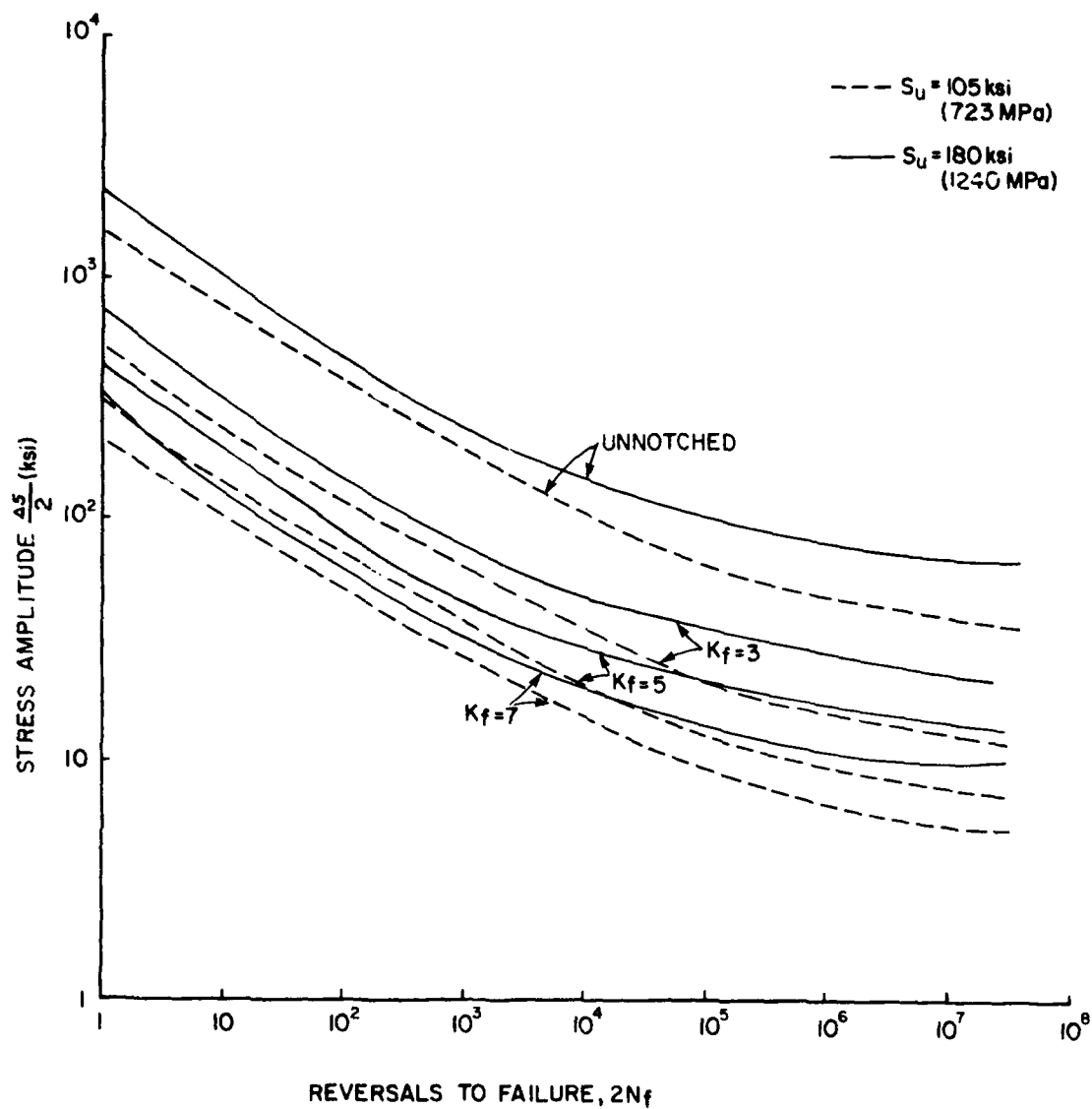


Figure 18. Notched specimen fatigue crack initiation life for various stress amplitudes. Unnotched specimen behavior is included for comparison with specimens containing fatigue strength reduction factors (K_f) of 3, 5, and 7. (The data shown are for a cyclic mean stress of 25 ksi [172 MPa].)

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